CHAPTER 14

Lubrication of Bearings With Liquid Metals

By WILLIAM J. ANDERSON

BECAUSE OF THE HIGH TEMPERATURE INVOLVED, closed-cycle space power systems will employ liquid metals and possibly other inorganic fluids as the cycle working fluid. One of the primary problems in space power systems is the rejection of heat. Since heat must be dissipated by thermal radiation, the surface area required to dissipate a given quantity of heat is inversely proportional to the fourth power of the radiating surface temperature. In order to keep the radiator area within reasonable limits, heat rejection temperatures must be kept high. Present theory is that this temperature will vary from about 400° to 1500° F. This temperature becomes the minimum cycle temperature unless auxiliary cooling, which is undesirable because of weight and complexity, is employed. Therefore, *minimum* bearing temperatures of 400° to 1500° F can be expected. Bearing temperatures will probably be somewhat higher than this, since the liquid leaving the radiator must be pumped to a high pressure and may be forced to flow through lines passing through hot areas. The use of organic fluids as lubricants at these temperature levels is not feasible.

In addition to avoiding problems of thermal degradation through the use of inorganic fluids as lubricants, using the cycle working fluid as the lubricant is highly advantageous because of several other reasons. These include

(1) Reduced complexity
(2) Ease of sealing problems
(3) Fewer mechanical design problems because of the elimination of overhung shafts, etc.

Fluids primarily considered for use as cycle working fluids are mercury and the alkali metals—rubidium, potassium, sodium, and

lithium. The properties of liquid metals which can affect the performance of bearings are their low viscosity and their corrosivity. The alkali metals will reduce most metal oxides, and the high mass density of mercury tends to promote erosion because of high particle inertia.

Because of the high chemical activity of the alkali metals, especially at elevated temperatures, materials compatibility is expected to be a severe problem. Materials can certainly be found which have good compatibility with the alkali metals; however, a good bearing material must also possess other characteristics.

SYMBOLS

C	clearance, in.
C_d	diametral clearance, in.
D	diameter, in.
f	coefficient of friction
L	length, in.
N	speed, rps
P	load pressure, W/LD, lb/sq in.
Re	Reynolds number, $\rho UC/\mu$
S	Sommerfeld number, $\left(\frac{D}{C_d}\right)^2 \frac{\mu N}{P}$
U	velocity, in./sec
W	load, lb
η	eccentricity ratio
μ	viscosity, (lb) (sec)/sq in.
ρ	mass density, (lb) $(sec^2)/in.^4$

TYPES OF BEARINGS

In evaluating the possible merits of various types of bearings for liquid-metal applications, it is necessary to consider the characteristics of each type of bearing in relation to the fluid properties. Rolling-element bearings, hydrodynamic bearings, and hydrostatic bearings will be considered separately.

Rolling-Element Bearings

In order for the rolling-element bearing to function properly, especially in high-speed applications, there must be no wear or surface degradation of the rolling elements or races. A high-speed rolling-element bearing is a precise machine component and must retain its precision during operation. Surface damage of any kind results in high-amplitude vibrations and rough operation, which further accel-

erate wear and quickly lead to complete failure. This is not true of the cage or separator, which can tolerate considerable wear. The combination of high pressures which exist in the contacts between the rolling elements and the races (even under light load conditions) plus the poor conformity of these contacting surfaces discourages the formation of a hydrodynamic film, which would prevent intimate contact of the surfaces. The success of any bearing in a liquid-metal environment will depend partly on preventing intimate contact of the surfaces. The liquid metal will either reduce the thin oxide coatings which exist on the bearing parts or prevent the reformation of the coating after it has worn off.

The use of surface coatings or contaminating films to reduce friction and wear is well known. In addition to low shear strength coatings that are used as lubricants, simple surface films, such as oxides, are valuable because they help to prevent clean surfaces from coming into intimate contact. In this way they are effective in preventing (or minimizing) surface welding. In the Earth's atmosphere, contaminating films of one type or another are present on all surfaces.

In a rolling bearing the net result of poor conformity is that clean metal surfaces come into intimate contact. This contact produces surface welding, which may result in material transfer or catastrophic wear, either or which constitutes bearing failure.

A further requirement of rolling-element bearings is an absolute compatibility of the race and the rolling-element materials with the environmental fluid for the same reason as the complete absence of wear. In other words, any corrosive surface damage that occurs on the rolling elements or races would have the same end effect as wear or surface damage arising from effects other than corrosion. This requirement for absolute material compatibility with the environmental fluid results in a low probability of success for rolling bearings in applications involving liquid metals where a long life with high reliability is required.

It is quite true that successful operation of rolling-element bearings has been achieved in other applications, such as in liquid hydrogen, where the lubricating ability of the environmental fluid is at best marginal. The temperature levels of liquid-metal applications plus the corrosivity of the fluids, however, would generally prevent the use of the self-lubricating retainer materials of the type which have been successful in cryogenic applications (e.g., Teflon). In addition, success in cryogenic applications has been achieved only because of the short life requirements. Bearings for chemical-rocket turbopumps, for instance, need have a life of no more than several hours. Bearings for auxiliary electric power systems for space vehicles will, in contrast, be required to operate reliably for 10,000 hours.

Hydrodynamic Bearings

A second type of bearing under consideration for liquid-metal applications is the hydrodynamic (fluid film) bearing. In this type of bearing, a continuous film maintains separation of the surfaces in relative sliding, and the pressure required to support the load is generated within the bearing itself. In a hydrodynamic bearing, the only external pressure required is that sufficient to feed only enough lubricant into the bearing to maintain a full fluid film. Hydrodynamic bearings offer several advantages when compared with rolling-element bearings in applications such as these. The first advantage is that of tolerating some wear and still maintaining their function. They can tolerate some corrosion and surface damage so that the requirements for material compatibility are not as severe as those for rolling-element bearings. The materials problem for hydrodynamic bearings is also less stringent because high hardness is not a requirement, whereas it is for rolling-element bearings. This requirement of high material hardness for rolling-element bearings eliminates a number of materials which might have excellent compatibility, but which are otherwise deficient with regard to hardness. The maintenance in hydrodynamic bearings of a fluid film (which keeps the surfaces in relative motion apart) tends to minimize the importance of the reduction of surface oxides and the resulting production of clean, nascent metals. As long as these materials are kept apart and prevented from coming into intimate contact, the possibilities of surface welding, material transfer, and high rates of wear occurring are minimized.

The principal disadvantage of hydrodynamic bearings lies in their tendency to exhibit instability under the light or zero load conditions that will exist in machinery in orbit. The term instability in hydrodynamic bearings refers to the tendency of the journal or shaft center to rotate in some orbit rather than to remain fixed in one spot within the bearing. If the amplitude of this journal center motion becomes large enough, touching of the shaft and the bearing occurs with, in most cases, a disastrous failure. This tendency toward instabilities in hydrodynamic bearings arises because the radial fluid film force is generally not colinear with the load vector. This produces a component of force, which tends to propel the journal in a whirling motion; hence, if the whirl is truly unstable, failure of the bearing follows. Whether or not the whirl is unstable depends upon the radial fluid-film stiffness. When the whirl frequency or the frequency of rotation of the journal center is equal to one-half that of the journal about its own center, the load-carrying capacity, or the radial-film stiffness, of a hydrodynamic bearing is essentially zero. Because of this relation, instability frequencies are most often equal to one-half

the frequency of rotation of the shaft. This type of self-excited whirl is commonly called half-frequency whirl. If a rotor is run at twice its first critical speed, a type of instability known as resonant whip develops. The whirl frequency is equal to one-half the rotational frequency (the resonant frequency of the rotor). Excessive amplitudes of vibration, which can lead to destruction of the bearings, develop.

Hydrostatic Bearings

The third type of bearing which might possibly be used in liquid-metal applications is the hydrostatic bearing. This is a fluid-film bearing in which complete separation of the surfaces in relative motion is maintained by a continuous fluid film. In contrast with a hydrodynamic bearing, however, the load-supporting pressures in a hydrostatic bearing are supplied from an external pressure source. The pressures that support the load are not generated within the bearing but are supplied through a flow restrictor of some type from an external source. The hydrostatic bearing offers the advantages of high load capacity at no rotation or at low rotative speeds and a load capacity which is independent of the fluid viscosity. The load capacity of a hydrodynamic bearing, in contrast, is proportional to the viscosity of the fluid; since the viscosities of liquid metals are quite low, the load capacities of hydrodynamic bearings using liquid metals are likewise quite low.

The hydrostatic bearing also has several disadvantages. The first of these is a dependence on an external pressure source. This dependence requires that a relatively high pressure source of the liquid metal be available at all times. The flow through a hydrostatic bearing, because of the high inlet pressure, is generally considerably higher than that of a hydrodynamic bearing. This makes the power loss (associated with the pump capacity required to supply the bearings) somewhat higher than that of a hydrodynamic bearing. To function properly, a hydrostatic bearing also requires a flow resistance of some type in the feed line between the high-pressure source and the inlet to the bearing. The restrictor may be a capillary tube or an orifice. Flow restrictors are generally sensitive to dirt and may be sensitive to erosion over a long period of time, especially a sharp-edge orifice in the case of a high-density fluid such as mercury. On the other hand, the hydrostatic bearing offers relative freedom from instabilities such as half-frequency whirl present in hydrodynamic bearings. Because it deflects under load in the direction of the load, the radial-fluid-film force, which opposes the load, is generally colinear with the load vector. There is, therefore, no tendency to promote a whirling or orbital motion of the journal or shaft center.

BEARING EXPERIMENTS

Rolling-Element Bearings

As mentioned previously, the materials-compatibility problem with rolling-element bearings will be severe, at best, and perhaps even insurmountable because of the requirements of this type of bearing. Reference 1 contains the results of experiments conducted with ball bearings operating in sodium-potassium alloy or sodium at a temperature of 250° F. Bearings of 50- to 140-millimeter bore size were run at 850 to 3600 rpm at thrust loads of 400 to 4300 pounds. Seven different metal alloys and a cotton-cloth-laminated phenolic material were evaluated as retainers. With all the metallic retainers, failure was due to severe wear of the retainer and metal transfer from the retainer to the balls and races. The phenolic showed better wear characteristics than the metallics, but such materials are not suitable for the temperature levels to which bearings in space power systems will be subjected.

Some work was done in reference 1 on the effect of speed on bearing life with 7320 size (100 mm bore) bearings with malleable iron retainers (fig. 14–1). An increase in speed from 850 to 3600 rpm resulted in a 90 percent decrease in life. The results of reference 1 indicate that the future for rolling bearings in liquid metals at extreme temperatures is not promising and that major development efforts in this area should be concentrated on fluid-film bearings.

Fluid-Film Bearings

The major portion of bearing work with liquid metals has been done with fluid-film bearings. The performance of tungsten carbide journal bearings in both NaK and sodium is described in reference 2. Test bearings were 2⅜ inches in diameter and in length. Experiments were conducted at temperatures from 100° to 600° F in NaK and 250° to 450° F in sodium at speeds of 1200 to 1700 rpm. These test conditions correspond to a maximum sliding velocity of approximately 18 feet per second.

In the experiments of reference 2, the bearings were wetted by exposing them to the liquid metal at 450° F for at least 8 hours before being tested. Runs were made with increasing load until a sharp increase in motor power, which indicated a rupture of the lubricant film, was observed. The liquid-metal-lubricated journal bearings operated with full film hydrodynamic lubrication. This was determined from the measurements of bearing torque, which agreed fairly well with existing theory. Figure 14–2(a) is a plot of friction variable as a function of Sommerfeld number for NaK lubricated bearings. These data show increasing friction with increasing temperature at high values of Sommerfeld number with the various curves con-

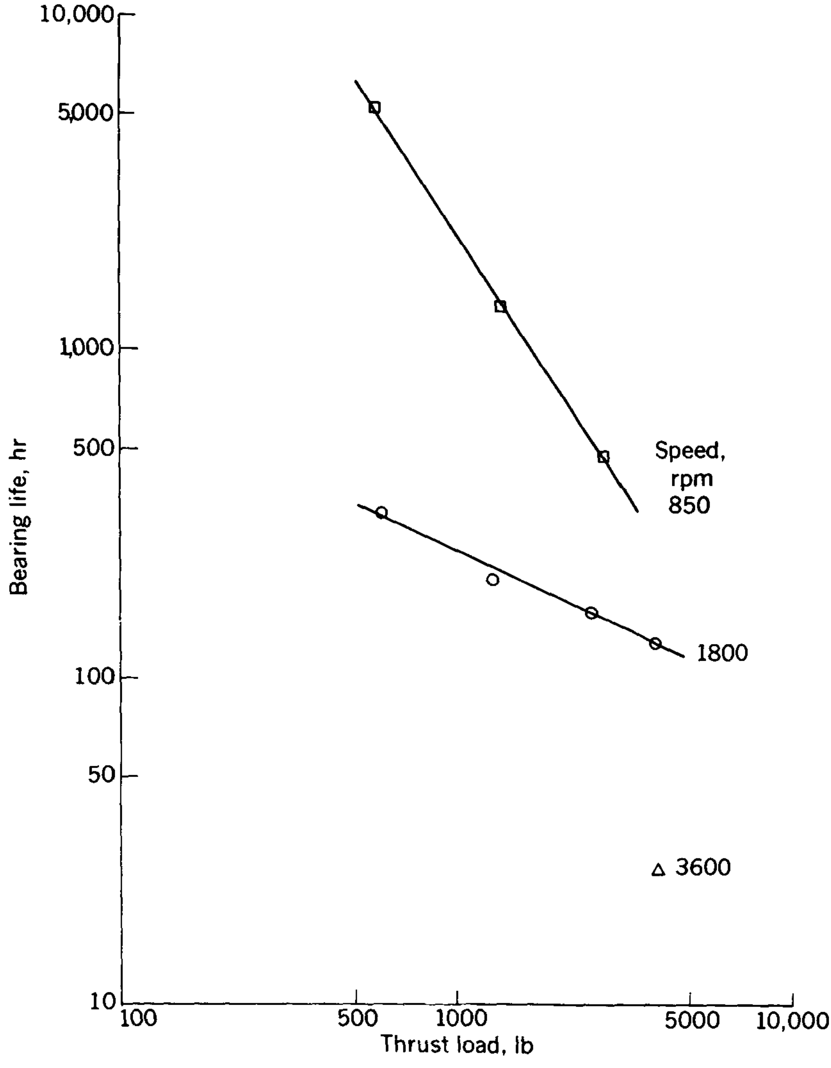
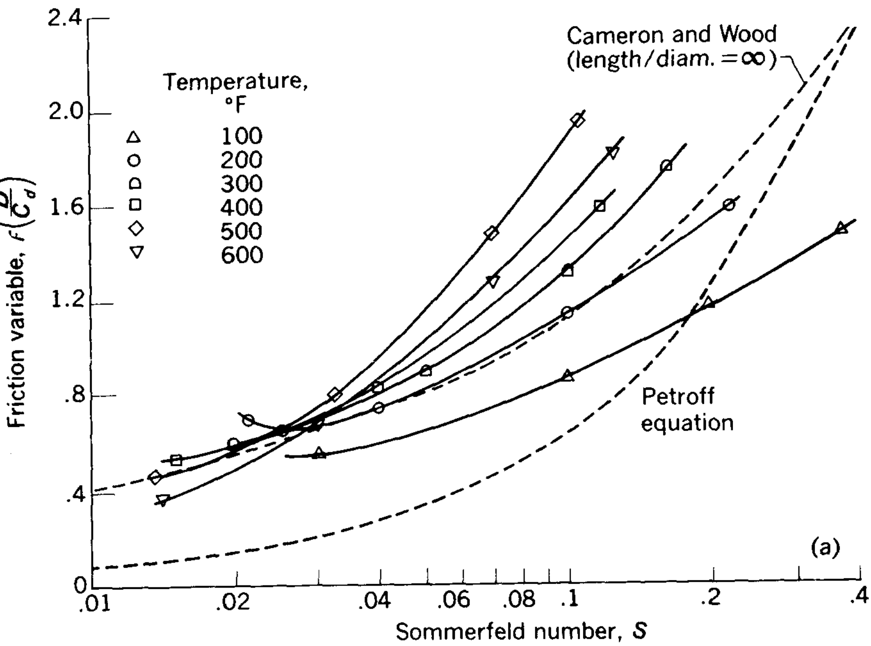


FIGURE 14-1.—Life of 7320 bearings at various speeds in sodium-potassium alloy at 250° F. Malleable iron ball guided retainers. (From ref. 1.)

verging to a single value of friction coefficient at a Sommerfeld number of 0.025. The curves of figure 14-2(a) were plotted by using the calculated bearing clearance at temperature based on known expansion coefficients of the bearing and journal materials. Any restraint on the bearing, however, may result in bearing operating clearances somewhat smaller than those predicted by using expansion coefficients and an initial clearance. This would shift the higher



(a) In sodium-potassium alloy at temperatures from 100° to 600° F.

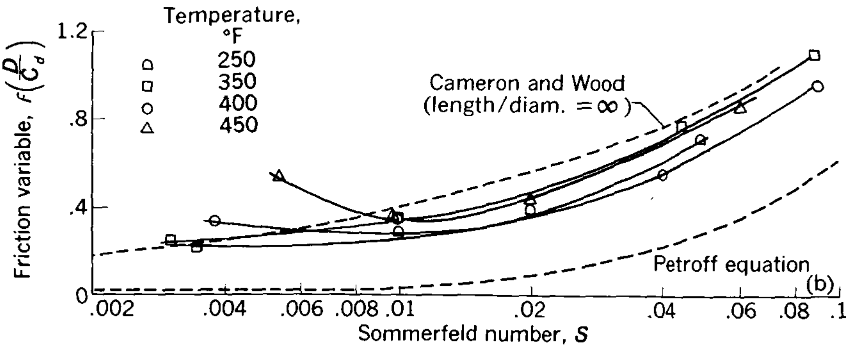
FIGURE 14–2.—Frictional characteristics of tungsten carbide journal bearings Test bearing, Carboloy 55A; test journal, Carboloy 779, bearing length-diameter ratio, 1; diameter, 2⅜ inches. (From ref. 2.)

temperature curves to the right and bring the curves into closer coincidence.

The author of reference 2 points out that higher temperatures may have produced misalinement of one of the bearings. Several of the high-temperature runs were discontinued because of roughening of the guide bearing surfaces due to boundary lubrication. Since the lubricant film thicknesses of the guide bearings should have been greater than those of the test bearings, it was concluded that misalinement of the guide bearings occurred at the higher temperatures.

Typical results obtained with sodium in reference 2 are shown in figure 14–2(b). The agreement between the data obtained at various temperatures is much better than in NaK, but the temperature range in sodium was narrower than in NaK. Examination of figure 14–2 shows that the frictional characteristics of bearings running in NaK and sodium can be predicted with fair accuracy using the Cameron and Wood analysis for an infinite bearing.

The test bearings used in reference 2 were Carboloy 55A (tungsten carbide plus 13 percent cobalt) and the test journals were Carboloy



(b) In sodium at temperatures from 250° to 450° F.

FIGURE 14-2—Concluded.—Frictional characteristics of tungsten carbide journal bearings. Test bearing, Carboloy 55A; test journal, Carboloy 779; bearing length-diameter ratio, 1; diameter, 2⅜ inches. (From ref. 2.)

779 (tungsten carbide plus 9 percent cobalt). Guide bearings were made of 18-4-1 tool steel or an aluminum alloy. The tungsten carbide cermets resisted seizure to a greater extent than did the other bearing materials, but some roughening of the bearing surfaces occurred for the particular ceramals used in the tests of reference 2.

Further discussions of the misalinement problem are given in reference 3, in which both thrust-bearing and journal-bearing tests are reported. These tests were conducted in NaK at temperatures from 75° to 760° F. Thrust bearing loads as high as 2000 pounds and journal bearing loads as high as 1800 pounds were carried. The journal bearings were tested at speeds from 900 to 1750 rpm and at loads in the region of 375 pounds per square inch. Boundary conditions occurred at high loads, and under conditions of film rupture tungsten carbide bearings showed better score resistance than tool-steel bearings. In addition to the materials used in reference 2, bearings were fabricated from Stellite Star J, Stellite 1, and Hastelloy B.

In reference 3, failure of the bearings to operate under boundary-lubrication conditions was attributed primarily to incompatibility of the bearing materials and the liquid metal. Many of the experiments reported in reference 3 are the same as those reported in reference 2.

Experiments with aluminum-alloy (percent composition: 4.4 copper, 0.8 silicon, 0.8 manganese, 0.8 magnesium, balance aluminum) thrust and journal bearings operating in NaK at temperatures of 80° to 400° F are reported in reference 4. Thrust bearings were 2¾ inches in outside diameter by 1¾ inches in inside diameter and were operated under a 75-pound load (21 lb/sq in.). They showed little deterioration

after 11,000 hours of exposure and intermittent operation in contaminated NaK. In contrast to the satisfactory performance of the thrust bearings, the 1-inch-diameter, 1-inch-long journal bearings failed under a 20-pound load (20 lb/sq in.). The initial journal-bearing diametral clearance was 0.002 inch, which was rebored to 0.006 inch because of the seizures.

The data of reference 4 indicate that the performance of aluminum-alloy bearings may be satisfactory in NaK at nominal temperatures if properly designed. For higher temperature use, the aluminum alloys may not be suitable for use in liquid metals because of poor compatibility.

Other data on the operation of fluid-film bearings at nominal speeds are given in references 5 and 6. It is apparent that the data reported in the literature on liquid-metal-lubricated bearings have been obtained at nominal temperatures and speeds. The anticipated bearing operating conditions in space powerplants range far beyond the experimental data available, both temperaturewise and speedwise. Therefore, the data discussed here can be used only as a guide to designing bearings for operation at more extreme conditions.

Some work on alkali-liquid-metal-lubricated fluid-film bearings operating at high speeds has recently been initiated. Experimental results for both water- and liquid-potassium-lubricated aluminum-bronze bearings are reported in reference 7. Journal bearings 1 inch in diameter were operated at speeds to 20,000 rpm in potassium at 450° F. Hydrodynamic operating conditions were achieved, and at the higher speeds significant increases in bearing torques, which indicate the presence of turbulent flow, were noted. This occurrence will be discussed in the section on turbulent flow.

The experimental results of reference 7 are quite promising. They indicate that operation of fluid-film bearings at the speeds required by space powerplants is feasible. Further work on bearing designs and materials is required to improve bearing reliability.

PROPERTIES OF BEARING MATERIALS

Compatibility

Certain conclusions can be drawn regarding material compatibility with NaK at 850° F. The results of corrosion tests conducted on several classes of materials are shown in table 14–I. From these tests the following conclusions were drawn:

(1) Porous tungsten carbide cermets gave excellent results and appear to be suitable for oscillating bearing applications.

(2) Combinations of titanium carbide and tungsten carbide cermets exhibit good compatibility and low coefficients of friction.

TABLE 14–I.—RESULTS OF FRICTION AND WEAR TESTS IN NaK AT 850° F [a]

[I, negligible wear; no material transfer; polish both specimens; II, smooth wear; fixed specimen; no material transfer; polish rotating specimen; III, smooth wear; both specimens; minor material transfer; IV, excessive wear; surface roughening; material transfer, or friction; V, test not completed because of excesses in IV.]

Specimen 1		Specimen 2													
Material	Composition	Carboloy 44A	Carboloy 779	Sintercast WF87	Carboloy X3040D	Carboloy 190	Carboloy 78	Carboloy 907	Carboloy 77B	Kentanium 138	Kentanium 138A	Firthite HT77	Colmonoy 6	Stellite 98M2	416 SS Chrome plated
Carboloy 44A	WC; Co binder	I	-----	I	-----	-----	-----	-----	-----	III	-----	III	-----	-----	-----
Carboloy X3040D		I	-----	-----	I	-----	-----	-----	-----	-----	-----	III	-----	-----	-----
Kentanium K-12		-----	-----	V	-----	-----	-----	-----	I	-----	-----	-----	III	-----	-----
Carboloy 55A		-----	-----	IV	-----	-----	-----	-----	-----	II	III	III	-----	-----	-----
Carboloy 190		-----	-----	-----	-----	III	-----	-----	-----	-----	-----	-----	-----	-----	-----
Carboloy 78		-----	-----	III	-----	-----	V	-----	-----	-----	-----	III	-----	-----	-----
Carboloy 907	WC; TiC, TaC; Co binder	-----	-----	I	-----	-----	-----	I	-----	-----	-----	III	-----	-----	-----
Carboloy 77B		-----	-----	-----	-----	-----	-----	-----	-----	-----	-----	III	-----	-----	-----
Carboloy 831		-----	-----	V	-----	-----	-----	-----	-----	-----	-----	I	-----	-----	-----
Kentanium 138	TiC; Co binder	-----	I	-----	I	-----	-----	-----	-----	III	III	-----	-----	-----	-----
Kentanium 138A		-----	I	-----	-----	-----	-----	-----	-----	V	III	-----	-----	IV	-----
Kentanium 151		-----	I	-----	-----	-----	-----	-----	-----	III	-----	I	-----	IV	-----
Kentanium 152B	TiC; Ni binder	-----	III	-----	-----	-----	-----	-----	-----	-----	I	-----	-----	-----	-----
Firthite HT77		-----	-----	-----	-----	-----	-----	-----	-----	-----	-----	III	-----	-----	-----
Kentanium	TiC; Fe binder	-----	I	-----	-----	-----	-----	-----	-----	-----	III	-----	-----	-----	V
Carboloy X3345	Cr_3C_2; Co binder	-----	-----	-----	-----	-----	III	-----	-----	-----	-----	-----	-----	-----	-----
Colomony 6		-----	IV	-----	-----	-----	-----	-----	-----	-----	-----	III	III	-----	-----
Stellite 98M2		-----	-----	-----	-----	-----	-----	-----	-----	-----	-----	-----	-----	IV	-----
Copper	Noncarbides	-----	-----	IV	-----	-----	-----	-----	-----	II	-----	-----	-----	-----	-----
Beryllium-copper		-----	-----	-----	-----	-----	-----	-----	-----	-----	-----	-----	-----	-----	II
Aluminum-bronze		-----	-----	-----	-----	-----	-----	-----	-----	-----	-----	-----	-----	III	-----

[a] From ref. 8.

(3) The lower the percentage of cobalt binder in tungsten carbide cermets, the less the tendency for superficial surface damage under boundary conditions. (This may be due either to the increased hardness or to the finer grain structure.)

(4) Titanium carbides not containing a solid-solution type of carbide exhibited less surface damage than did those containing solid-solution carbides.

(5) Titanium carbide cermets tested with similar cermets exhibited good compatibility at temperatures below 850° F.

(6) The compatibility of nickel-bonded cermets compares favorably with the compatibility of cobalt-bonded cermets.

(7) Copper and some copper alloys are most compatible with chromium or cermets at temperatures up to 600° F.

(8) Nickel and nickel alloys did not exhibit good compatibility at high temperatures.

(9) Chromium—tungsten-cobalt alloys have poor compatibility at temperatures above 600° F.

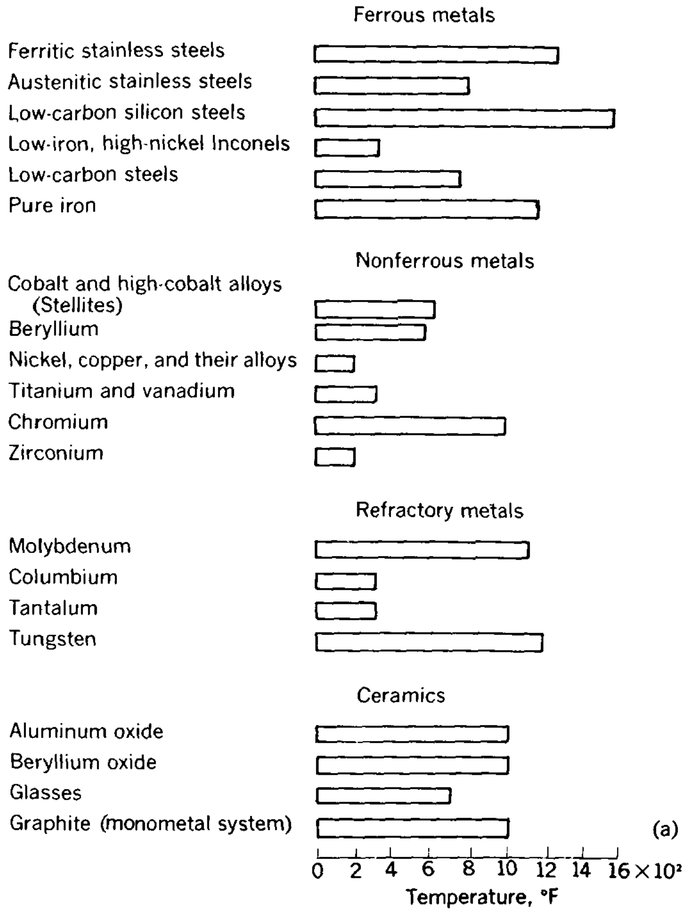
(10) Chromium carbide cermets are unsatisfactory bearing materials.

(11) Iron alloys are unsatisfactory at temperatures above 600° F.

More recent compatibility data have been compiled from a number of sources and are reported in reference 9. Compatibility ratings are based on a 10,000-hour exposure. These data are summarized in figure 14–3(a) for mercury, figure 14–3(b) for rubidium, figure 14–3(c) for sodium and potassium, and figure 14–3(d) for lithium. All these data were obtained in static tests with satisfactory compatibility based on an absence of corrosive attack.

Figure 14–3(a) indicates that the ferritic stainless steels and the low-carbon silicon steels have excellent compatibility with liquid mercury; they are satisfactory for use in dynamic systems to 1300° and 1500° F, respectively. The use of chromium, molybdenum, and silicon as alloying agents appears to be beneficial. It is advantageous in bearings to use a wetting agent. Magnesium offers promise in this area. It is to be noted from figure 14–3(a) that the materials discussed have better compatibility with liquid mercury than do the more expensive refractory metals and ceramics.

Figure 14–3(b) shows the compatibility of various classes of materials with liquid rubidium. The refractory metals—molybdenum, columbium, tantalum, and tungsten—are all compatible with rubidium to the maximum temperature of the investigation, 2000° F. The broken bars do not indicate the limit of compatibility but rather that no information is available on behavior at higher temperatures. Of the more conventional ferrous alloys, the ferritic stainless steels and the austenitic stainless steels are compatible to a temperature

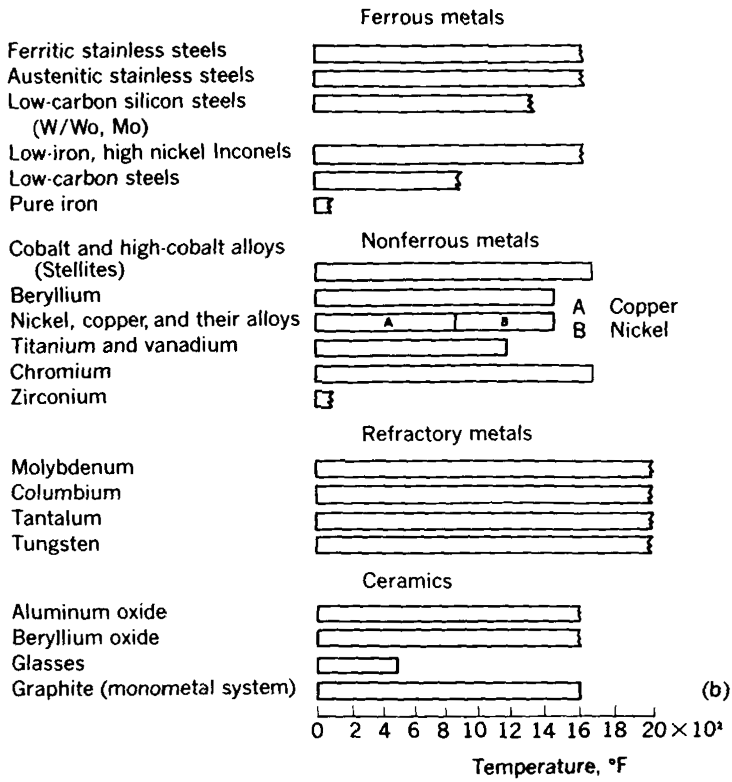


(a) Mercury.

FIGURE 14–3.—Compatibility of various materials with several liquid metals. Exposure, 10,000 hours. (From ref. 9.)

of at least 1600° F. The high-nickel alloys such as Inconel also exhibit good compatibility. Several of the 18–8 type austenitic stainless steels such as 310, 316, and 347 are suitable to the region of 1500° F. At higher temperatures the refractory metals are recommended.

The compatibility of various materials with liquid sodium and potassium is shown in figure 14–3(c). In general, the behavior of the various classes of materials with sodium and potassium is similar to that with rubidium. Both the austenitic stainless steels and the ferritic stainless as well as nickel alloys such as Inconel are satisfactory for use at temperatures to the region of 1500° F. As with rubidium,



(b) Rubidium.

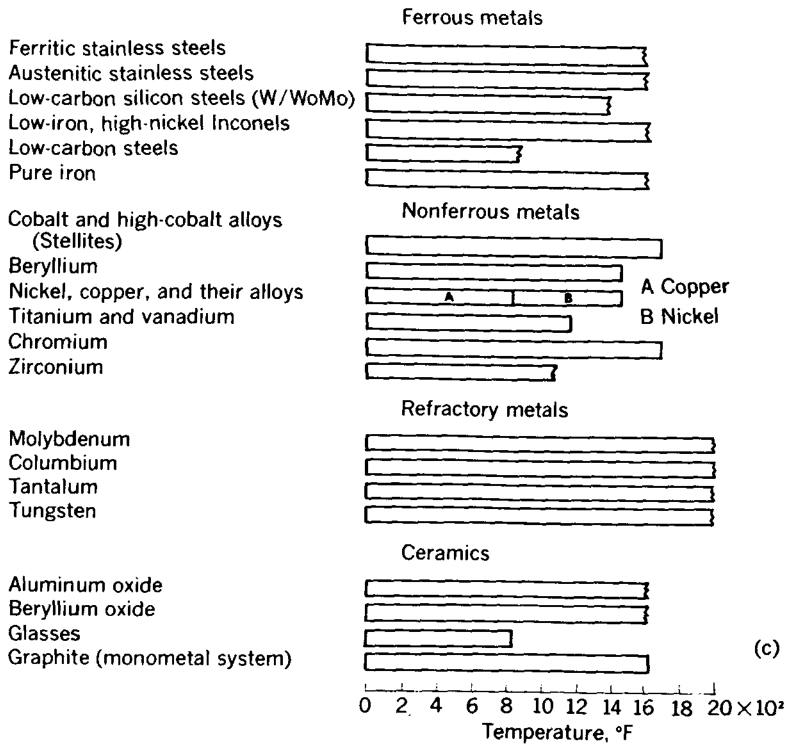
FIGURE 14–3—Continued.—Compatibility of various materials with several liquid metals. Exposure, 10,000 hours. (From ref. 9.)

the refractory metals are recommended for use at temperatures above 1500° F.

The most difficult of the alkali metals, from a materials viewpoint, is lithium. Conventional ferrous alloys and a variety of nonferrous alloys have satisfactory compatibility at temperatures from 600° to 1000° F. In particular, ferritic stainless steels and austenitic stainless steels may be used in dynamic systems at temperatures to 1000° F. At temperatures above 1000° F, the refractory metals must be used.

Static corrosion resistances of some alloys and elements in sodium at 1832° F (ref. 10) are shown in table 14–II. An alloy of 80 percent nickel and 20 percent chromium and pure nickel were the only materials unattacked after 400 hours of exposure.

Reference 11 contains data on the corrosion resistance of various ceramics and cermets to sodium and lithium. The data of reference 11 are based on a 100-hour exposure at 1500° F unless otherwise indicated.



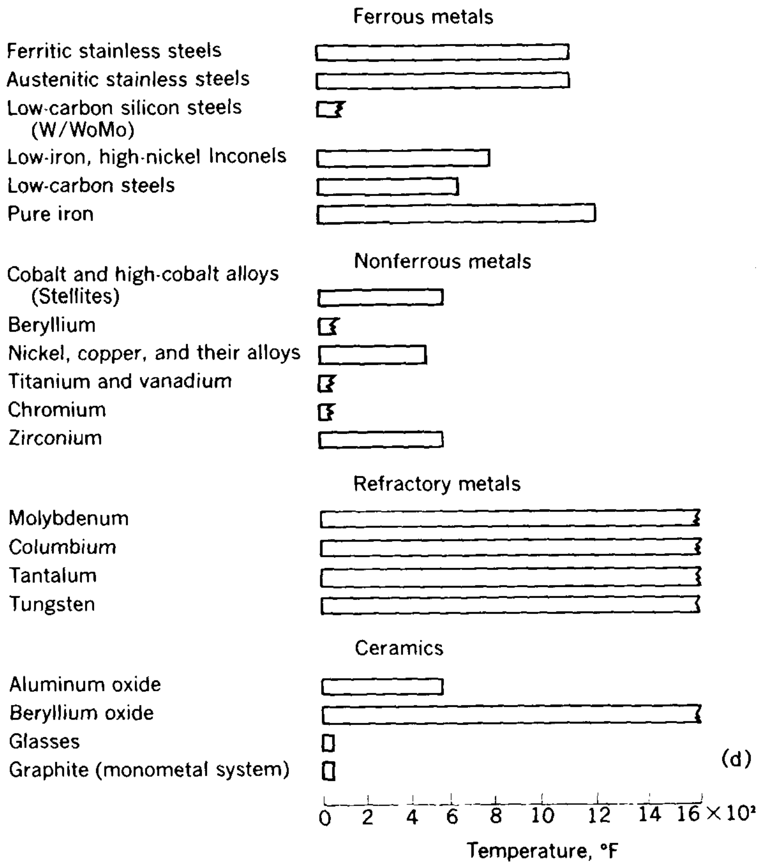
(c) Sodium and potassium.

FIGURE 14-3—Continued.—Compatibility of various materials with several liquid metals. Exposure, 10,000 hours. (From ref. 9.)

The data for ceramics in sodium and lithium are shown in table 14-III and the data for cermets in sodium in table 14-IV. As shown in table 14-III, a number of ceramics have good corrosion resistance in sodium. These include carbides, oxides, and borides. In the test of reference 11, good compatibility is defined as a depth of attack of 0.001 inch or less. As one would expect, the choice of materials for lithium is considerably narrower than for sodium. The carbides of titanium, zirconium, and chromium have good compatibility with lithium at 1500° F.

Among the cermet materials tested in reference 11, several of the Carboloys and Kentanium materials had good compatibility with sodium. None of these were tested in lithium.

The resistance to corrosion in liquid sodium and in lithium of the various ceramics tested in reference 11 generally agreed with the results that would be predicted for these media and ceramic materials from their free energies of formation. The author of reference 11, however, states that thermodynamic data are not available for many



(d) Lithium.

FIGURE 14-3—Concluded.—Compatibility of various materials with several liquid metals. Exposure, 10,000 hours. (From ref. 9.)

of the materials. The limited results obtained on single-crystal specimens tended to parallel predicted corrosion resistance trends, while the results obtained on polycrystalline sintered specimens did not. The author of reference 11 concludes that thermodynamic data can be useful as qualitative guides for planning tests and for selecting materials to be tested, but corrosion tests are required to secure the quantitative data.

As discussed in chapter 13, material incompatibility may manifest itself as mass transfer. Material is transferred from a high-temperature zone to a low-temperature zone because of differences in solubility in the alkali metal. This phenomenon is more likely to occur in a system containing several materials, and therefore it would be advantageous to design a monometallic system. This system, however,

TABLE 14–II.—STATIC CORROSION RESISTANCE OF ELEMENTS AND ALLOYS IN SODIUM [a]

[Temperature, 1832° F; 400 hr.]

	Unattacked	Very good	Good	Fair
Alloys	80 percent Ni; 20 percent Cr	446 SS [b] 316 SS Inconel Stellite 25 Vitallium 310 SS	410 SS 430 SS [b] 446 SS [c] 304 SS 347 SS	405 SS
Elements	Nickel	Titanium Chromium Cobalt Molybdenum Tantalum	Beryllium	Vanadium

[a] From ref. 10.
[b] 0.08 percent C.
[c] 0.006 percent C.

TABLE 14–III.—CERAMIC MATERIALS HAVING GOOD CORROSION RESISTANCE IN SODIUM AND LITHIUM AT 1500° F [a]

[100-hr test unless otherwise indicated.]

Sodium	Lithium
ZrB_2 TiC ZrC Cr_3C_2 BeO MgO (single crystal) Al_2O_3 (single crystal) Sm_2O_3 (1000-hr test) Rare-earth oxides body (45 to 49.5 percent Sm_2O_3 plus 22.5 to 27 percent Gd_2O_3; balance primarily other rare-earth oxides). $MgAl_2O_4$ (single crystal)	TiC ZrC Cr_3C_2

[a] From ref. 11.

may not be feasible because materials with desirable structural properties may not have the properties desired in a bearing material. A material may be chosen for system piping, etc., because it possesses good compatibility and is easily machined and welded.

Wear and Seizure Resistance

In addition to compatibility, which (in the case of bearing materials for liquid metals) implies a resistance to corrosion, fluid-film bearing materials should possess other characteristics. First in importance after compatibility is wear and seizure resistance. This cannot be considered a property of either the bearing or the journal material,

TABLE 14-IV.—CERMET MATERIALS HAVING GOOD CORROSION RESISTANCE IN SODIUM [a]
[100-hr test.]

Material	Test conditions
Carboloy 779 (91 percent WC—9 percent Co) Carboloy 55A (87 percent WC—13 percent Co) Carboloy 907 (74 percent WC—20 percent TaC—6 percent Co) Carboloy 608 (83 percent Cr_3C_2—2 percent WC—15 percent Ni)	Static at 1500° F
Kentanium 150A (80 percent TiC—10 percent $NbTaTiC_3$—10 percent Ni) Kentanium K151A (70 percent TiC—10 percent $NbTaTiC_3$—20 percent Ni)	Seesaw test: Hot zone, 1500° F Cold zone, 1150° F

[a] From ref. 11.

but one of both the bearing and the journal materials in combination with the particular liquid metal used. The necessity for good wear resistance is self-evident since the wear that a bearing can tolerate and function properly is limited. Of somewhat less importance than wear resistance is low friction. High resistance to wear and low friction do not, in many instances, occur simultaneously. If a fluid-film bearing is operating in the hydrodynamic region, the bearing friction is a function of the viscosity of the lubricant rather than of the coefficient of sliding friction between the bearing and the journal materials.

Embeddability

At the extreme condition of temperature under which liquid-metal-lubricated bearings will operate, some wear and surface degradation can be expected. Wear particles will therefore be present in the bearing. In addition, other solids such as oxides coming out of solution may be flowing through the bearing. These situations emphasize the need for bearing materials with good embeddability, which is the ability to absorb foreign particles and thus minimize scoring and wear. The experimental evidence obtained to date indicates that hard materials such as tungsten carbide perform best in liquid metals. The fact that this class of materials has poor embeddability emphasizes the need for extremely good wear resistance.

Conformability

Conformability, the ability to compensate for misalinement and geometric inaccuracies, is another important property of bearing materials. The problem of misalinement will most certainly be present

in liquid-metals systems because of the extreme temperatures. Good conformability generally results from low hardness and low modulus of elasticity. Unfortunately, the classes of materials which exhibit good compatibility with liquid metals generally have high hardness and high modulus of elasticity. The necessity for maintaining extreme cleanliness and accuracy in liquid-metal systems is apparent because of the probable poor embeddability and conformability of bearing materials. The necessity for research on different classes of materials with low hardness and low modulus of elasticity is also indicated.

Low-Shear-Strength Surface Films

The importance of contaminating films in reducing friction and wear was discussed thoroughly in chapter 2. In liquid-metals systems, such films may be of greater importance than in oil-lubricated systems because of the inherently poor lubricating qualities of the liquid metals and the poor friction and wear characteristics of materials compatible with the liquid metals. The formation of one such film, sodium molybdate, on molybdenum in sodium was investigated in references 12 and 13. The experiments conducted in references 12 and 13 were low-speed friction tests at sliding speeds of about ½ inch per minute, a Hertz contact stress of about 100,000 psi, and temperatures of 80° to 1300° F. With tungsten carbide (20 percent cobalt binder) in sodium (ref. 12), kinetic friction remained high and essentially constant up to 1000° F and increased above 1000° F. Molybdenum, however, showed considerably lower friction in the presence of sodium at higher temperatures. Moreover, marked changes in friction occurred when the molybdenum specimens were conditioned by exposure to various temperatures for extended periods in vacuum or in inert-gas atmospheres. These results were attributed to the formation of the oxide molybdenum trioxide (MoO_3), which was formed when molybdenum was heated in air, and to molybdenum dioxide (MoO_2), which was formed after heating in vacuum. The coefficient of friction for the MoO_2 was less than for the MoO_3, but surface damage was comparable and relatively severe. When sodium was added to the MoO_2 filmed specimens, the friction was somewhat lower up to 1000° F and surface damage was essentially nonexistent. In this case, the surface-film composition was sodium molybdate (Na_2MoO_4). The results of friction tests of filmed molybdenum specimens run dry and in sodium are shown in figure 14–4.

From the experiments of reference 12, it is evident that the formation of sodium molybdate films is highly beneficial because of their low friction and good resistance to wear and surface damage at high unit loads. Some oxygen must be present to form this film in a practical sodium system. As stated by the authors of reference 12,

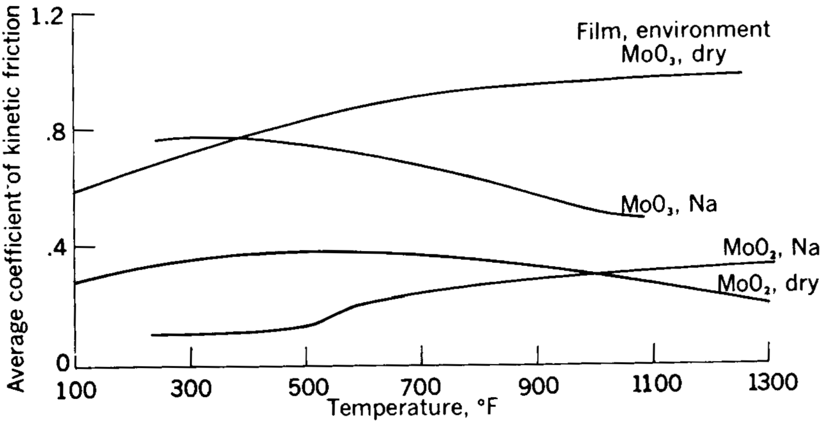


FIGURE 14-4.—Summary of molybdenum friction studies with respect to different surface films found on test surfaces. (From ref. 12.)

the oxygen may be initially available as sodium oxide (Na_2O), but this may be removed by filters. The use of preformed films on sliding bearing surfaces may be necessary.

Since tungsten behaves chemically like molybdenum, the authors of reference 13 investigated the possibility of the formation of sodium tungstate films (Na_2WO_4) on sodium-lubricated tungsten specimens. The coefficients of friction for dry and sodium-lubricated tungsten are shown in figure 14-5. As with molybdenum, friction when the specimen was run in sodium was lower than when it was run dry. The initial high friction at temperatures below 600° F suggests that

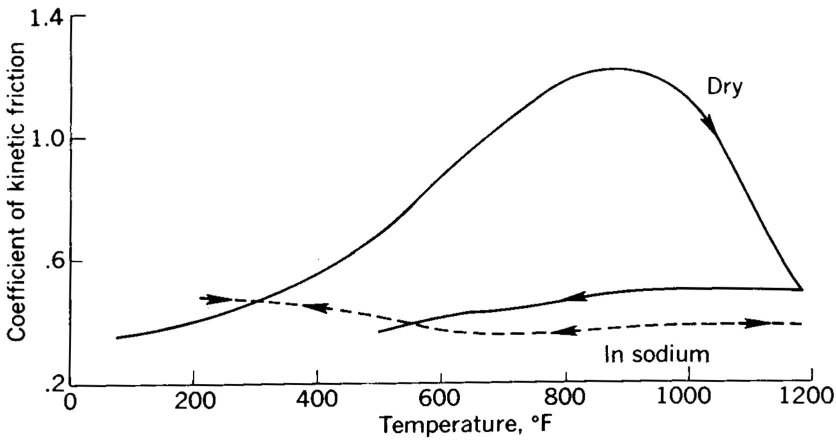
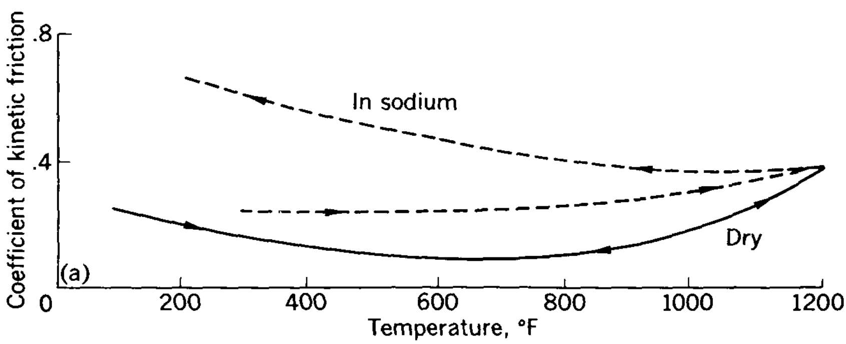


FIGURE 14-5.—Friction-temperature behavior for tungsten-tungsten system. (From ref. 13.)

the sodium tungstate film may not have formed until subsequent exposure at higher temperatures. The presence of a sodium tungstate film was verified by electron-diffraction techniques on a specimen exposed to sodium vapor at 1200° F. Both molybdate and tungstate films afforded protection to their respective substrate materials equally well under the test conditions of references 12 and 13.

The frictional behavior of a titanium carbide—niobium carbide—cobalt (TiC—NbC—Co) cermet was also investigated dry and in sodium (ref. 13). The behavior of the titanium carbide cermet was quite different from that of molybdenum and tungsten in that the cermet exhibited lower friction when run dry than when run in sodium. These results are shown in figure 14–6(a). The relatively low kinetic



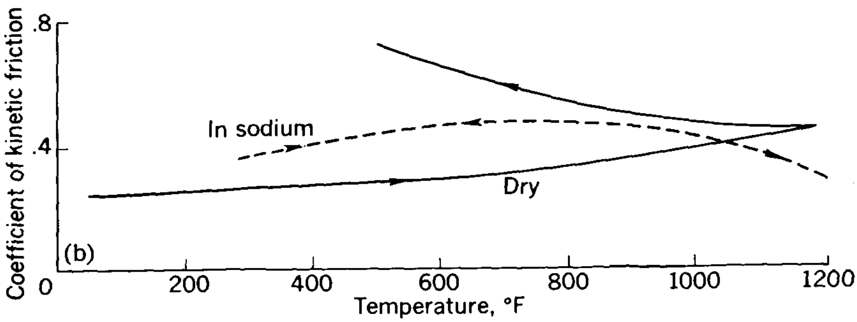
(a) Titanium carbide—titanium carbide system.

FIGURE 14–6.—Friction-temperature behavior of two systems. (From ref. 13.)

friction under dry conditions is believed to be due to the presence of a preformed film of titanium dioxide. In sodium it was postulated that a sodium titanate film was formed, although residual sodium on the surface prevented positive identification.

In reference 13 the frictional behavior of a tungsten carbide—cobalt (WC—Co) cermet was found to be similar to that of the TiC—NbC—Co cermet. These results are shown in figure 14–6(b). The addition of sodium stabilized the frictional level for both ascending and descending temperatures but did not produce significantly lower values of friction. Films similar to those postulated for tungsten may have been formed on the WC, but the differences in frictional behavior may be associated with the hardness of the substrate.

The overall results obtained in references 12 and 13 indicate that the presence of surface films on substrates can provide effective protection against gross surface damage. Caution must be used in extrapolating these results to high-velocity systems because the experiments were conducted in a stick-slip apparatus at a very low sliding



(b) Tungsten carbide—tungsten carbide system.

FIGURE 14-6—Concluded.—Friction-temperature behavior of two systems. (From ref. 13.)

velocity. It must also be kept in mind that the formation of such films in practical systems may be very difficult.

PROPERTIES OF LIQUID METALS

The most important property of a lubricant in fluid-film bearings is viscosity. The load-carrying capacity of hydrodynamic bearings is directly proportional to the lubricant viscosity, while the flow from the bearing (the flow required to maintain a film) is inversely proportional to viscosity. Thus the low viscosities of the liquid metals result in low load capacity and high flow requirements. On the other hand, bearing power loss is proportional to viscosity so that the power loss in liquid-metal bearings should not be of concern unless turbulent flow occurs. One of the problems concerned with the use of liquid metals will, therefore, be that of low load capacity, while the advantage gained in reducing power loss is not expected to be significant. Viscosities of four alkali metals and mercury are shown in figure 14-7. Values are given at temperatures up to the boiling point for all the fluids except lithium. At approximately 200° F the viscosities of the alkali metals are on the order of 10^{-7} reyn. At this temperature an SAE 30 oil has a viscosity of about 20×10^{-7} reyn. Thus, relative to a medium viscosity mineral oil, reductions in load capacity greater than an order of magnitude can be expected with the alkali liquid metals.

Other properties of interest, including melting and boiling points, thermal conductivity, specific heat, density, and viscosity of liquid and vapor phases, are given in table 14-V. Relative to conventional oils, the liquid metals have lower specific heats and higher thermal conductivities. A lower specific heat will result in a higher temperature rise for a given amount of shear work done on the fluid. On

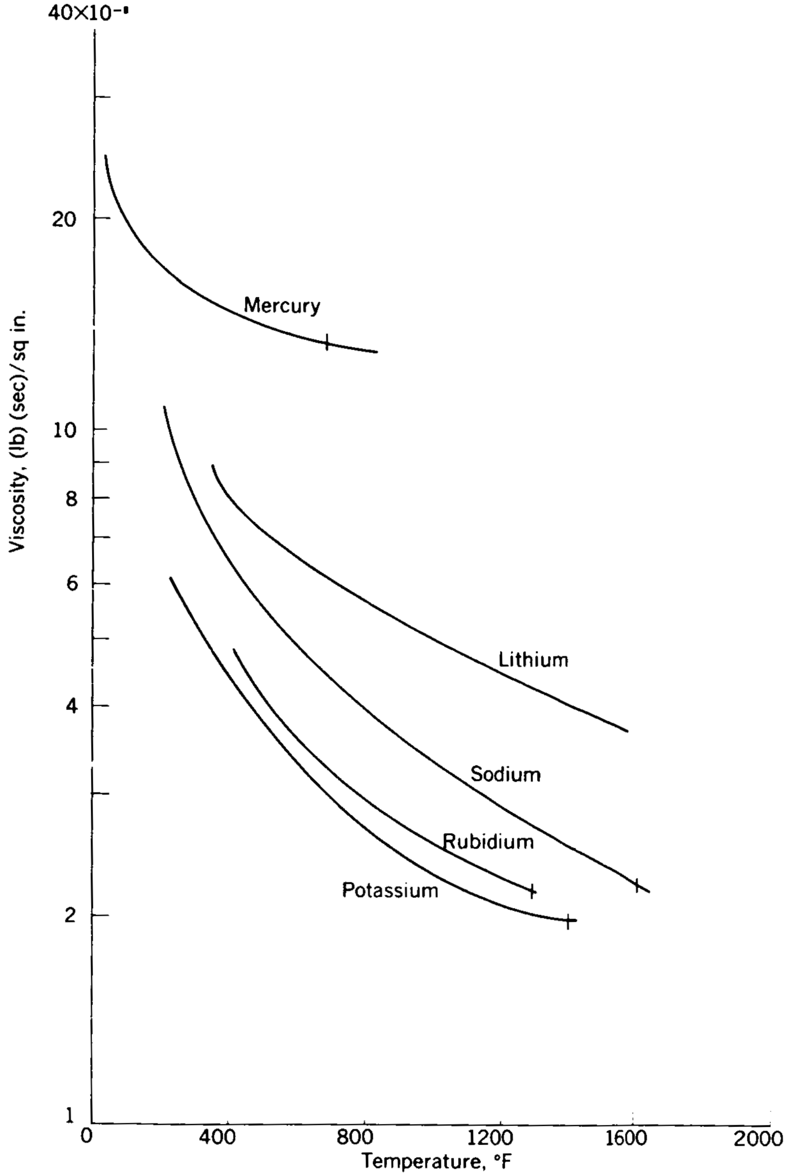


FIGURE 14-7.—Viscosity of mercury and four alkali liquid metals. (From ref. 9.)

the other hand, a higher thermal conductivity will result in lower temperature gradients within the fluid for a given quantity of heat transferred. The net result will be that liquid-metal-lubricated bearings will experience a different thermal growth from convention-

TABLE 14-V.—PROPERTIES OF INORGANIC FLUIDS FOR SPACE APPLICATIONS [a]

	Metal				
	Mercury	Rubidium	Potassium	Sodium	Lithium
Melting point, ° F	−38	102	145.	208	357
Boiling point, ° F	674	1270	1400	1618	2437
Density of liquid at boiling point, lb/cu ft	794	82. 15	41. 4	46. 3	26. 2
Density of vapor at boiling point, lb/cu ft	0. 26	0. 060	0. 0305	0. 0165	------------
Viscosity of liquid at boiling point, (lb) (sec)/sq in.	1.34×10^{-7}	2.17×10^{-8}	1.94×10^{-8}	2.21×10^{-8}	8.98×10^{-8} [b]
Viscosity of vapor at boiling point, (lb) (sec)/sq in.	8.85×10^{-9}	2.43×10^{-9}	1.41×10^{-9}	1.87×10^{-9}	------------
Surface tension, lb/ft, at ° F	0. 0318 at 68	0. 0034 at 102	0. 0059 at 145. 8	0. 01306 at 212	0. 027 at 357
Vapor pressure at 2200° R, lb/sq in. abs	3000 (extrapolated)	112	65	26. 5	5
Thermal conductivity of liquid, Btu/(hr) (ft) (° F)	8. 1 [c]	11. 8 [c]	18. 0 [c]	30. 1 [c]	27. 2 [b]
Thermal conductivity of vapor, Btu/(hr) (ft) (° F)	0. 0055 [c]	0. 00354 [c]	0. 0045 [c]	0. 0083 [c]	------------
Specific heat of liquid, Btu/(lb) (° F)	0. 0323 [c]	0. 0877	0. 1868 [c]	0. 305 [c]	0. 98 [c]
Specific heat of vapor, Btu/(lb) (° F)	0. 0248 [c]	0. 0579 [c]	0. 1267 [c]	0. 214 [c]	0. 716

[a] From ref. 9.
[b] At melting point.
[c] At boiling point.

ally lubricated bearings. This factor must be considered in setting bearing clearances.

The specific heat of the fluid will determine its ability to carry away heat since the product of specific heat, temperature rise, and flow equals the rate of heat dissipation. The low specific heat of the liquid metals may determine the minimum allowable flow. The probable high operating temperature of liquid-metal-lubricated bearings will tend to minimize this problem, however.

Liquid-metal-lubricated bearings operating in the region of the fluid boiling point may be subject to cavitation. If the pressure at any point in the bearing falls below the vapor pressure of the lubricant, vaporization can occur with a resultant discontinuity in the fluid film. Cavitation forces can promote erosion of the bearing material.

OPERATING PROBLEMS

Space powerplants will be called upon to operate unattended for long periods of time at high rotative speeds in a weightless environment. The low viscosity of the working fluid will result in low bearing load capacity, and the combination of low viscosity and high speed will help to promote turbulent flow. Bearing power consumption under turbulent flow conditions is significantly higher than with laminar flow. The weightless environment intensifies problems related to bearing instability. Material problems such as compatibility and dimensional stability are aggravated by the extreme temperatures to which the bearings will be subjected. All these problems with the exception of those arising from bearing instability and turbulence have been discussed previously. These two problems will now be discussed in detail.

The origin of half-frequency whirl has been discussed previously in chapter 3. In space powerplants, the occurrence of this type of whirl would be encouraged by the high speeds, low viscosity, and absence of gravity forces. In systems where whirl is a problem, it may be necessary to resort to the use of nonstandard journal-bearing configurations. Several types of bearings with good anti-whirl characteristics have been designed into rotating machinery. These are, in approximate order of increasing ability to prevent whirl,

(1) Elliptical bearings
(2) Pressure bearings
(3) Longitudinal groove bearings
(4) Three-lobe bearings
(5) Pivoted-shoe bearings
(6) Nutcracker-type bearings

All these bearings suppress whirl by generating an artificial load within the bearing. This artificial load is created by generating

wedge films, by dragging fluid into a pressure pool with restricted flow paths, or by tapping fluid from the high-pressure region and feeding it back into the bearing on the unloaded side. Determination of the operating characteristics of and design procedures for antiwhirl bearing shapes is at best difficult and sometimes impossible unless many simplifying assumptions are made. Solutions must often be obtained by trial and error because of the interaction of the various regions of the bearing. For example, in a three-lobe bearing, the operating characteristics of each lobe can be computed by considering each lobe to be a 120°-arc journal bearing. The forces generated in each lobe, however, are interdependent so that the solution must be sought by trial-and-error methods until the summation of all the oil film forces is equal and opposite to the external bearing load. Procedures for determining the characteristics of several bearing types are given in reference 14.

Turbulence

Most fluid-film-bearing analyses are based on the assumption of laminar flow. These analyses are valid so long as the speed at which turbulent flow begins is not exceeded. The transition to turbulent flow begins when the Reynolds number is given by the following equation (ref. 14, p. 231):

$$\frac{\pi \rho N D C_d}{2\mu} = 41.1\sqrt{\frac{D}{C_d}} \tag{14–1}$$

When turbulent flow exists, bearing power loss rises and the oil flow decreases. Figure 8–13, on page 231 of reference 14 is a chart for computing the power loss under turbulent flow conditions. The experiments of reference 6 showed that the transition from laminar to turbulent flow occurred at higher speeds than predicted by Taylor's criterion (eq. (14–1)).

An investigation of the load-carrying capacity of journal bearings with turbulent flow was carried out by Tao (ref. 15). Tao assumed that the short-bearing-theory assumption (that the pressure flow in the circumferential direction is small compared with the velocity flow) is valid and that the pressure flow in the z-, or axial, direction can be derived from the Blasius one-seventh power law of velocity profile.

The results of Tao's analysis, plotted as eccentricity ratio against dimensionless numbers, are shown in figures 14–8 and 14–9. Figure 14–9 shows that the load capacity of a bearing operating with turbulent flow is greater than that of a bearing operating with laminar flow. Thus, turbulent flow results in increased load capacity and friction and decreased flow. In reference 7, recorded bearing torques in the turbulent region were 4 to 40 times as great as predicted by laminar theory.

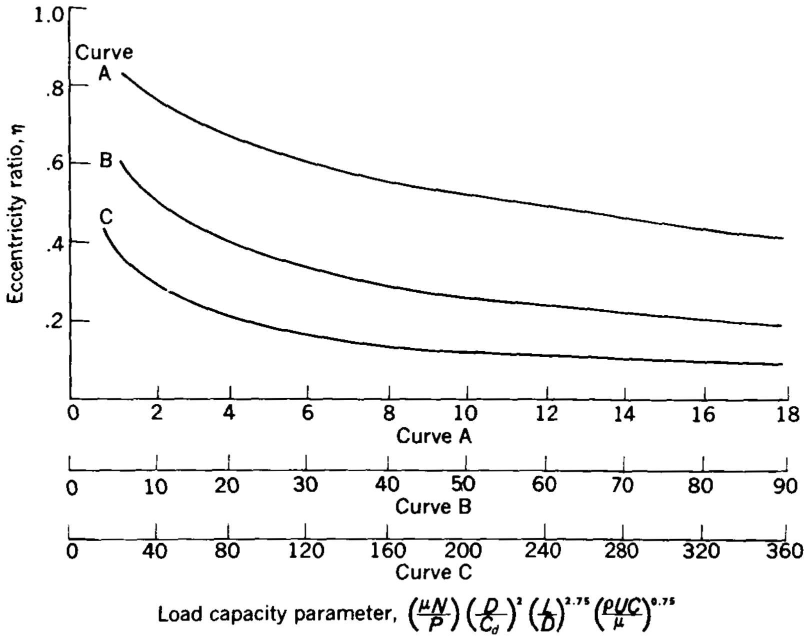


FIGURE 14-8.—Eccentricity ratio as function of load-capacity parameter for fully developed turbulent flow. (From ref. 15.)

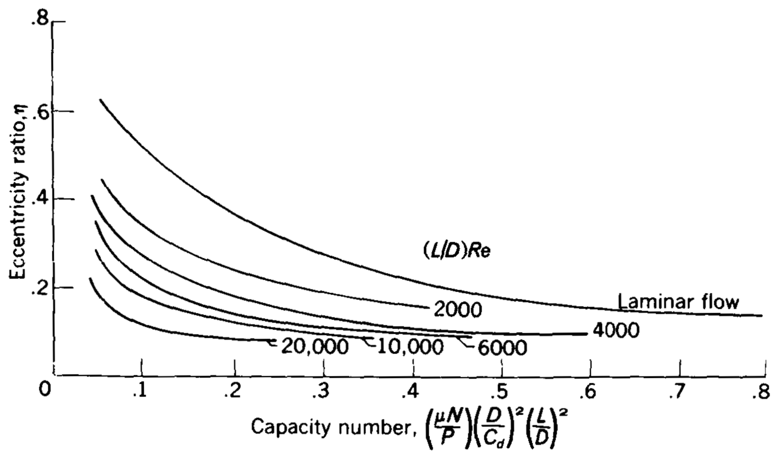


FIGURE 14-9.—Eccentricity ratio as function of capacity number for laminar flow and turbulent flow at various Reynolds numbers. (From ref. 15.)

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